

AVIATION AND AERONAUTICAL ENGINEERING



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SEPTEMBER

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SEPTEMBER 1, 1916

AVIATION AND AERONAUTICAL ENGINEERING

VOL. I. NO. 3

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Its construction for service

This machine was tested by Victor Carlstrom on August 21 before the Naval Armament Board, consisting of Naval Constructor R. C. Richardson, U. S. N., Lt. Col. W. G. Child, U. S. N., Bureau of Steam Engineering, and Lt. Col. C. B. Brown, U. S. N. It passed six tests, climbing 1,000 feet in ten minutes, attained a maximum speed of seventy-five miles an hour with a maximum of forty-five and left the water in 150 yards.

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MARTINUS BECHTOLD
EDWARD W. WILLIAMS, R. E.

Vol. 1

September 1, 1918

No. 3

On August 22, for the first time in the history of the United States regular army, aeroplanes passed in review of their commanding officer with troops of other arms. This review was held "somewhere in Mexico" before Brigadier-General John J. Pershing. Long should this epoch making date be remembered and great should be the praise of those making the event possible. But equally interesting is the fact that the machines were efficient for combat service.

The twelve aeroplanes with General Pershing's expedition are all equipped with engines of 100 horsepower or over and have machine guns mounted for immediate service. Every aeroplanist carries also two .303 Winchester automatic rifles with 200 rounds of ammunition. Besides of both an incendiary chamber and also explosive bombs form part of the equipment and these are automatic weapons with this squadron. After all the emphasis that has been laid at the aviation arm of the service, it is a genuine pleasure to make a record of which every American should be proud.

And the performance of these machines during July, gave promise of a brilliant future for the aeronautical squadron. These twelve machines were in the air a total of three hundred and sixty hours, about two hours a day for a month. This thing was accomplished under the most unfavorable conditions and proves that the many difficulties which were encountered at the outset of the campaign have been overcome.

The problem of suitable propellers has been partially solved by the invention of a propeller factory on the border. Propellers are never used for two consecutive weeks, experiments in succession. As soon as a machine returns to its base at Columbus, N. M., the propeller is removed and placed in a humidor to preserve the viscosity of the glue which holds together the humidor and another propeller is taken out and attached to the machine when the time comes for its first flight. Experiments with two, three, four and five-bladed propellers are being conducted.

The propeller problem will also be greatly minimized by giving down the motor, which is also being done on new machines. Experiments with propellers of steel, and plated blades are being conducted by several important companies. The much thought cannot be given to this essential factor.

This revolution in the affairs of the aero squadron under General Pershing is chiefly due to the untiring work of Lieut.-Col. George O. Squier, U. S. A., head of the aviation section of the signal corps. He brought to his present assignment a knowledge of conditions abroad

and a determination to put aviation in the army on a sound basis. He has done it magnificently. All honor to him.

General Kitchener and "Every aeroplanist is worth an army corps." It would be interesting to know the opinion of General Pershing as to the value of an aeroplanist to him. As a cavalry officer of the highest reputation, knowing the extreme importance of reconnaissance, he must have found in the serviceable aerial equipment furnished him a splendid opportunity for reconnaissance. It is to be hoped that he will sometime give the country his opinion.

Dep Control Now Standard in Army and Navy

By a recent order of the War Department it is provided that all army aeroplanes are to be equipped with Dependant or Dep control system. These will replace the controls now in use as soon as the army there can convert themselves to the new controls. Two training machines have been ordered for the aviation base at the border, equipped with the old control in one and the Dep in the other. By using these machines it is expected that the army there can rapidly be taught the Dep system.

Dep control is used exclusively by the Royal Flying Corps of Great Britain and the French flying corps. It has become the standard control used on all types of machines in Europe. Lateral stability is maintained by one of the wheel mounted on a steering column. This column moves forward and back and by this movement the elevating planes are manipulated. Steering is accomplished by pressing with either foot on a rudder bar, the right foot being pressed forward to turn the machine to the right and vice versa.

In the old control system lateral stability was maintained by the pilot swinging his body to right or left, thus moving a shoulder yoke which controlled the wing-warping devices. Army pilots complained that this system compelled them to maintain an unstable position and also interfered with them when they wished to look over the side of the fuselage to observe the terrain immediately beneath their machines.

Until recently the Navy used the so-called modified Dep control in which the wing warp and elevator were controlled as in the Dep, but the steering bar wires were crossed so that the right foot was pressed forward to turn the machine to the left. The Navy Department has followed the army's lead in ordering that Dep control to still exclusively on all navy planes.

The design of rubber shock absorbers or springs for aeroplane landing carriages is very unsatisfactory from an engineering standpoint because of uncertainty with regard to the mechanical properties of the rubber. Rubber is in almost universal use for medium and light weight aeroplanes and there appears to be no other material so well adapted to this particular employment.

It is notorious that landing carriages are rarely broken by rough landings, and rule of thumb design gives no idea of the extent of such failures nor the factor of safety provided. Consequently, I have endeavored to compile data upon the mechanical properties of such rubber as is commonly used on aeroplanes, and to apply the results to design.

It appears that the commercial grade of rubber springs on the market is not in every way what one would desire. The elongation permissible is small, which allows only a small travel for the landing gear so that the energy of landing must be absorbed. The shock to the aeroplane is accordingly very severe and it is remarkable that there are not more accidents.

Rubber Manufacture*

Processes of manufacture include the breaking down and washing of the crude rubber, compounding or mixing with retinators rubber or shoddy, rubber substitution, sulphur and mineral fillers, etc. The effect upon the mechanical properties of the finished product of the various compounding materials is the technique of the manufacturer.

R. H. Vance of the Goodyear Tire & Rubber Co. has been so kind as to give me the following classification of the compounding materials in common use:

"(1) Crude Rubber, including hard, medium, soft and very soft, also Goeta Percha, Goeta Sisk and Balata. Rubber forms the basis of all good quality rubber goods. The kind or quality of the rubber determines to a large extent the quality of the cured finished article.

"(2) Retinators rubber; used mainly as a cheapener or filler. The composition of a retinator and the proportion which it has to a tendency to impart to a stock, depend largely upon the kind of scrap from which it is obtained.

"(3) Rubber substitutes, used mainly as light gravity fillers.

"(4) Bitumen, tann, gutta, mineral rubber, and asphalt, are all of similar nature. They are used mainly as cheap binders, or to facilitate the processing of the stock, such as mixing and extruding.

"(5) Resins, used mainly as cheapeners and fillers and exert a hardening effect on the dry natural particles.

"(6) Waxes, cerates, paraffins, beeswax, caribwax, and for special purposes such as for obtaining a gas tight film and to obtain special results in hard rubber.

"(7) Fats and oily retinators of, rosin oil, castor oil, paraffin oil, and vasoline, used mainly to facilitate mixing and handling of the stock.

"(8) Mineral powders, this is such a large class that it should be considered under the following sub-classes, although many of these sub-classes are of the properties of two or more of these sub-classes.

"(9) Vulcanizing agents, which include sulphur as the only

one commonly used. Sulphur Chloride is used in vapor or liquid form.

"(10) Accelerators, which hasten and modify the curing rate of rubber and sulphur, including lead, magnesium salt, ammonium carbonate, azobenzene, sulphide, telluride and many other lead compounds.

"(11) Liquid processing, some powders are added for the special properties which they give the compound. The most important is zinc oxide, with its peculiar toughening properties. Oleic acid and lamp black also belong to this class.

"(12) Cheapeners or fillers, including whiting, barites, lime and clay.

"(13) Pigments, which are used principally for the color other than black, including lithopone, E. M. blue, English red, iron yellow, lamp black, iron oxide, etc."

In the process of vulcanizing, rubber stock which sulphur has been mixed, is heated to a temperature above the melting point of sulphur. The rubber and combined sulphur make a very elastic compound of much greater strength and elasticity which is furthermore less affected by temperature changes than the original crude rubber. The chemical formula for rubber gum is not yet known with certainty but is of the form $(C_4H_6)_n$, and each group C_4H_6 carries with two atoms of sulphur when vulcanized.

There is no standard method for the chemical analysis of rubber. It is usual to remove mechanical impurities and wash soluble matter by washing. Then the sulphur, mineral salt, wax and resin are extracted with acetone. The sulphur content of the extract is found by ordinary methods, and the solid residue "organic extract" or "unvulcanized acetone extract" is weighed. This residue matter should not exceed 5 per cent of the rubber in the compound for high grade material.

Mechanical Properties of Rubber in General

In general, we may expect that with the very best grades of rubber and the utmost skill in compounding and vulcanizing the tensile strength of rubber may be about 2000 lb. per sq. in. Such rubber may be stretched to about six times its original length. Ordinary commercial rubbers do not approach these figures.

In any rubber the stress-strain curve is not a straight line and hence there is no proper modulus of elasticity, as is the case with metals below their elastic limit. However, for engineering purposes it is achieved in rough calculations to approximate the general shape of the stress-strain curve assumed to be a straight line. In the design of aeroplane landing carriages an extension of less than 500 per cent is contemplated for the rubber, and even this may give the modulus of elasticity on the slope of the above curve does not change rapidly.

Rubber differs from the metals in having a large "hysteresis factor." This is when a rubber band has been stretched and the value of load and elongation plotted it is found as the load is gradually removed that the rubber does not contract so rapidly as before and the return curve makes with the first curve a "hysteresis loop" representing work done on the rubber which is not returned. This work loss represents the shock absorbing quality of the rubber. The work lost in hysteresis in low grade heavily compounded rubbers may be as much as 70 per cent of the work done upon first elongation. Better grade rubbers show a hysteresis loss of about 40 per cent.

It would appear that the pure grade rubber had the advantage as a shock absorber. However, under repetition of the

last cycle the hysteresis factor for the heavily compounded rubbers appears that for the pure gum. Furthermore, no cheap stock does not stretch to our use as well as the other.

After stretching and release, heavily compounded rubber has a large permanent set which will in time nearly disappear. The pure gum stock has less set, recovers more quickly and has a more stable characteristic curve.

On an aeroplane landing carriage, the rubber taken up the first shock of landing on a suddenly applied load which is quickly returned. On the first cycle the hysteresis loop will be large and the aeroplane will be thrown back with only about half the energy with which it stretched the rubber. Releasing the ground, a rapid extension of bumps will cause the motion to go through stress-strain cycles with such rapidity that there is no time for recovery of the permanent set given by the first cycle. The hysteresis factor may now be only 20 or 30 per cent. This is exactly what we desirably changed spring to take up the minor bumps caused by irregularities in the ground.

Methods of Testing

As there is no standard method for chemical analysis of rubber, so there is no standard method for mechanical testing. Frequently a small straight strip is pulled in the jaws of a testing machine. The rate at which the load is applied and released, number of cycles, etc., have an important effect upon the results.

For aeroplanes, rubber is most commonly employed in the form of a loop or ring. Tests made on a ring show a lower tensile strength than tensile tests made on a straight strip of

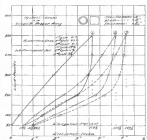


FIG. 1. HYSTERESIS LOOPS FOR GRADE A RUBBER RINGS

the same rubber, due to the difference in tension between the static and moving layers of the ring and to change in extension. In view of our use on an aeroplane rubber rings should give better figures for use in design.

Tests at the Bureau of Standards (for ref.) showed that the rate of loading has only a secondary effect upon the results of the test. Change of temperature showed a substantial more effect. An increase from 50 deg F to 90 deg F showed an

average 16 per cent decrease in tensile strength, 36 per cent increase in elongation and 30 per cent decrease in permanent set.

In some tests made at the Massachusetts Institute of Technology by Mr. Hafl, the actual rubber rings as supplied to aeroplane builders were shipped over 15-m. bolts (the one used on the aeroplane) stretched to the limit of an ordinary wire testing machine. Temperature for all tests was about

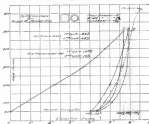


FIG. 2. HYSTERESIS LOOPS FOR GRADE B RUBBER RINGS

(in fig. 2, the load was applied slowly, about 3 in. in 30 seconds).

Elongation and Tensile Strength

Four specimens of "Wright type" rubber rings, grade A, mean diameter 2 in., width 2 in., thickness 5/16 in., broke at an average of 900 lb. per sq. in. and showed an average ultimate elongation of 265 per cent of the initial length. The tensile strength was reduced as the half mean perimeter. The tensile strength is computed on a basis of the original cross-sectional area.

Two specimens of the same type and grade of rubber which had been kept in an office for one year showed a tensile strength of 750 lb. per sq. in. and ultimate elongation of 240 per cent. The effect of age seems to be not very marked.

Tests on two specimens of "Bureau type" rings, grade A, mean diameter 2.5 in., width 1 in., thickness 1/8 in., gave an average value 600 lb. per sq. in. and 300 per cent elongation.

Two rings, 2 in. x 2 in. x 5/16 in., made of another grade B of rubber (slightly inferior) showed a tensile strength of 135 lb. per sq. in. and an ultimate elongation of only 200 per cent. These three rings could only be tested in length when rupture occurred. To all external appearance these poor rings were no different from the lot first tested.

To the aeroplane designer it is very necessary to know how far the rubbers may safely be allowed to stretch. It will be noted that the best of these rings do not approach the tensile strength of 2000 lb. per sq. in. and an elongation of 500 per cent which we assume good rubber can be made to show.

Hysteresis Loss

The curve of Fig. 3 shows the stress-strain curve for one of the "Wright type" rings for three cycles. The ones were measured with no time interval to get the exact permanent set.

* HUNSAKER, CHAPTER No. 25 of the Bureau of Standards, The Properties of Rubber, 1926, 1927, 1928, 1929, 1930, 1931, 1932, 1933, 1934, 1935, 1936, 1937, 1938, 1939, 1940, 1941, 1942, 1943, 1944, 1945, 1946, 1947, 1948, 1949, 1950, 1951, 1952, 1953, 1954, 1955, 1956, 1957, 1958, 1959, 1960, 1961, 1962, 1963, 1964, 1965, 1966, 1967, 1968, 1969, 1970, 1971, 1972, 1973, 1974, 1975, 1976, 1977, 1978, 1979, 1980, 1981, 1982, 1983, 1984, 1985, 1986, 1987, 1988, 1989, 1990, 1991, 1992, 1993, 1994, 1995, 1996, 1997, 1998, 1999, 2000.

that given it, without waiting more than three minutes for more complete recovery from stress. The hysteresis loss, or ratio of area of loop to area under ascending stress curve is becoming smaller and from 51 per cent for the first cycle is only 41 per cent for the third cycle. The sub-permanent set is almost entirely given by the first cycle and is not great.

Similar hysteresis curves were found in many other specimens, but without material difference in character. The hysteresis losses given above may be taken as representative of commercial acrolein supplies. Builders of this grade are



FIG. 1. BRIDGE TYPE SHOCK ABSORBER READY FOR TEST

in general use in the United States and appear to be fully satisfactory for the lighter airplanes.

Fig. 2 shows the hysteresis loops for the samples of B grade rubber mentioned above in connection with the tests on ultimate strength and elongation. The curve of the first cycle is somewhat starting in appearance, giving a hysteresis loss of about 50 per cent. The sub-permanent set after the first cycle is relatively enormous. For the first blow, the rubber should be an ideal shock absorber were it not for the low tensile strength and small allowable elongation. As a matter of fact, the rubber from which these craps were made was originally developed for balloons as railroad equipment, where only compression loads are considered. It is not a suitable grade for tension work.

Modulus of Elasticity

As mentioned above, there is no real modulus of elasticity for rubber, but we find useful an approximate figure representing the slope of a straight line which roughly represents the ascending curve of the stress-strain diagram. We have here taken the ascending curve of the first cycle for the computation of E in the expression:

$$f = E \frac{x}{100}$$

where f is ultimate stress in lb. per sq. in., E is modulus of elasticity and x is ultimate elongation in per cent of original length.

From tests on Grade A rubber craps, we find E for four specimens 359, 346, 316, 283 lb. per sq. in. The Grade B rubber, two specimens gave $E = 275$ and 258. For Grade A rubber we may take $E = 330$ as a safe figure. If we compute E for the part of the stress-strain diagram below 50 per cent of breaking load we find an average value of about 290 lb. per sq. in.

Test of Complete Shock Absorber

A bridge type shock absorber made for an actual airplane was tested by L. B. Q. Jones and H. A. Harrow, U. S. Army. The general arrangement is shown in Figs. 2, 4, 5. There was fitted 32 rubber strips 2 in. x 2 in. x 5-10 in., similar to that described above in connection with Grade A rubber. The ring was made over a $\frac{3}{4}$ -in. steel pin.

The laboratory report of the test follows:

Department of Mechanical Engineering,
Massachusetts Institute of Technology,
Testing Materials Laboratory, 200 St. St.
HYPERSTRESS TEST OF AIRCRAFT SHOCK ABSORBERS, AND THE
FOR EXCESSIVE STRENGTH OF SAME.

Specimen

Air-Gliding, 2 in. in diameter by $\frac{3}{4}$ in. thick.
Bridge and plate, and rolled steel, $\frac{1}{4}$ in. thick.
Rubber—Twelve 2 in. by 2-5-10 in. commercial rubber craps.
Pin— $\frac{3}{4}$ -in. steel.

The Tests

The elongations at the various loadings were obtained by taking the distance (average) between four pairs of semi-permanent marks.

Three runs for hysteresis were made as follows:

First Elongation in.	Second Elongation in.	Third Elongation in.	Fourth Elongation in.	Fifth Elongation in.	
1.000	2.2	1.000	4.0	1.000	8.2
1.200	4.0	1.200	1.20	2.000	1.7
2.000	7.0	2.000	1.20	2.000	1.7
2.000	9.10	2.000	1.90	2.000	5.12
1.000	1.40	1.000	1.90	2.000	5.12
2.500	2.70	2.500	2.70	2.500	2.70
4.000	2.00	4.000	2.00	4.000	2.00
5.500	2.00	5.500	2.00	5.500	2.00
1.000	1.80	1.000	4.00	1.000	1.40
2.000	5.70	2.000	5.70	2.000	1.90
2.000	2.20	2.000	5.70	2.000	1.90
1.000	2.10	1.000	2.10	1.000	2.10
2.000	6.70	2.000	6.70	2.000	6.70
2.000	5.10	2.000	5.10	2.000	5.10

At the beginning of the test run a total load of 300 lb. was applied between 1 and 2 in. of the pin marks was 2.5 in. A fourth run was then made for ultimate strength. At a load of 8750 lb. with a corresponding elongation of 5.60 in., the lower bridge failed by buckling under the air, and no pin at that point failed by shearing off on the inside of the

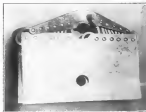


FIG. 4. THE LANDING GEAR BRIDGE WITH TWELVE RUBBER STRIPS.

plate. At the time of the failure there was a pronounced slip of the rubber, but none of the rubber failed.
(Signed) B. Q. Jones,
H. A. Harrow.

Discussion of Results

Fig. 6 is a plot of the stress cycle of the hysteresis run. The hysteresis loss drops from 47 per cent to 38 per cent, as would be expected. The sub-permanent set is practically constant at about 39 per cent after the first cycle. For the specimen under the load was run up each time to 45 per cent of the load which ultimately broke the bridge. At this point, 400 lb. the stress in the rubber was only 197 lb. per sq. in., which appears to account in a qualitative way for a lower



FIG. 5. DETAIL OF CONSTRUCTION OF LANDING GEAR BRIDGE UNDER TEST

ultimate loss than we had found for the individual rubbers also caused by a higher stress. In this connection it is of interest to observe that the hysteresis loss is greater for greater elongations.

The initial length of an unstretched rubber is taken as the half span perimeter or 21.4 in. The bridge failed at a load of 8750 lb., at which point the rubbers had stretched 560 per cent more than the original length and were stressed 550 lb. per sq. in. From previous tests these rubbers were found to break at about 580 lb. per sq. in.

The modulus of elasticity computed from $\frac{100f}{x} = \frac{E}{100}$ = 68 lb. per sq. in. This is somewhat greater than the highest value, 508, put from tests on individual rubbers. Here again the discrepancy comes from the fact that the rubbers in the two tests were not equally stressed.

The buckling of the bridge before the rubbers were fully elongated shows the landing gear in service might collapse in load shock. Ordinarily a movement of 5 in. for the shock absorbing mechanism is small.

An airplane weighing 10 pounds striking the ground at 1 ft per second as a glide of 1 in. 7 has kinetic energy to be absorbed by the landing gear of $\frac{WV^2}{2g}$. If the machine were to rest after a motion of a foot, the work done by friction is in Wx , and the total energy stored in the shock absorber is $W \left(x + \frac{1}{2g} \left(\frac{V^2}{g} \right) \right)$. The average force in the springs is half the maximum P_1 given by the equality:

$$\frac{1}{2} P_1 x = W \left(x + \frac{1}{2g} \left(\frac{V^2}{g} \right) \right)$$

$$P = W \left(2 + \frac{1}{2g} \left(\frac{V^2}{g} \right) \right)$$

If we take ordinary conditions as $V = 55$ ft. per sec. (45 mph), $P = W \left(2 + \frac{1}{2g} \left(\frac{V^2}{g} \right) \right)$, from which we get the following table for use in design:

x	P
1 inch	68 W
3	23 W
4	37 W
5	53 W
6	71 W
8	96 W
10	124 W
12	157 W

It appears that the load on the landing gear is nearly 34 times the weight of the airplane, if a motion of only 5 in. is allowed. This requires an excessive factor of safety and makes a very heavy construction. Of course, an allowance has been made for the collapse of permanent stress which may add 3 in. to the motion of the vessel mechanism.

For our particular gear, failure comes at 8750 lb. for a motion of 5 in. This gear then should not be used on airplanes weighing more than $\frac{8750}{34} = 257$ lb. total if they are expected to land at so steep a glide and at a speed of 45 miles per hour. The gear is of the type used today on 2000-lb. airplanes, and it is clear why their landing requires such definite handling.

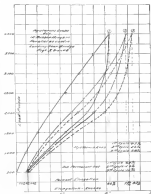


FIG. 6. HYSTERESIS LOOPS FOR TWELVE RUBBER STRIPS IN PARALLEL AS USED IN LANDING GEAR BRIDGE.

FIG. 3, 4 AND 5.

The bridge under test failed by buckling due to insufficient bending between the webs. This is purely a fault in design and for a very slight increase in weight a bridge of similar type can be made which will develop the full strength of the

are mounted. This appears to be a very happy feature of the design.

Figs. 2 and 3 illustrate the general arrangement very well. The cylinders are cast singly. A double carburetor is used, one for each set of three cylinders, and two magneto. The main shaft is connected with the vertical shaft. The upper vertical shaft drives the magneto, the main

Crank-shaft and Crank-case

The crank-shaft and crank-case are shown in Figs. 6, 7 and 8. The crank-shaft rests of course on main bearings, its cylinders being mounted separately. It is very strong and

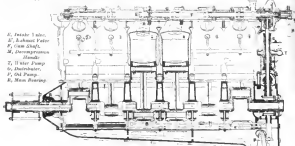


FIG. 5. LONGITUDINAL SECTION OF THE 360 H. P. MERIDIAN MOTOR

vertical air pump and the horizontal overhead shaft, which is turn is responsible for the exhaust and inlet runs of the overhead valves and for the air pump. The lower vertical shaft is connected to the oil pump which feeds oil into the crank case under pressure.

Cylinders, Valves, Pistons and Connecting Rods

The cylinders, as shown in Figs. 4 and 5, are of cast steel, forged for machining, machined everywhere, with welded sheet steel jackets, and automatically welded valve seatings. They are very strong and comparatively light for the single-cylinder type of construction. Complete with springs, valves, water inlet and two plugs, a single cylinder weighs 22.5 pounds.

The valves offer no particular features for comment. The pistons present nothing striking. The cylindrical part of the piston is of cast iron, the rest of steel. The piston rings are 3.57 inch in diameter and 0.185 inch high. The total length of the piston is 4.75 inches, and the total weight, together with rings, gudgeon pin and fitting screws, is 0.8 pounds. The gudgeon pin is hollow and of very large diameter, 1.20 inches inside and 1 inch outside, with a slightly tapered section.

Between the steel gudgeon pin and the small end of the connecting rod is a cast iron pin, perforated in about eight places to permit the free passage of oil and to maintain a vacuum. The connecting rod, of S-form, is very strong in spite of its comparative lightness (3.1 pounds). The large end is lightened by perforating the hole and cap

weights 78.5 pounds complete, with dimensions of 21.5 inches for the main and 2.2 inches for the bearings. The rod bearings are somewhat longer than the intermediate ones.

There are two oil reservoirs, one at each end of the crank-shaft, which are interconnected by the pump and a pipe of fairly large diameter and covered by fully perforated plates, as in Fig. 6. The oil circulating system will maintain its work in whatever be the inclination of the screw-pump. The valve of the aluminum crank-case are very thin, but a very rapid construction is obtained

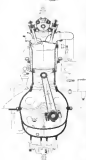


FIG. 6. LOWER HALF OF CARBURATOR, SHOWING REINFORCED DOWN PIPES



FIG. 7. LOWER HALF OF CRANK CASE, SHOWING AIR OVERFLOW FOR CARBURATOR

its internal reinforcement.

From Fig. 4 it will be seen that the crank-case has a series of double bottom, forming a strapping, while the fin of the same drawing project under its whole length and, as well as adding considerably to the strength, serve to reduce heat from the oil reservoir. The partition shown is also fixed in the bottom portion of the crank-case. Materially lightened, they do not add much to the weight, but help to strengthen the bearings very considerably.

Cooling System

The centrifugal water pump is also driven from the upper vertical shaft, as shown in Fig. 2. A defect of the cooling system is that the cooled water



FIG. 8. LOWER HALF OF CRANK CASE, SHOWING AIR OVERFLOW FOR CARBURATOR

Carburetor

The carburetor, as mentioned above, is double. It is made of sheet aluminum, and the mixing chamber is surrounded by a jacket, through which hot water passes on its way from the cylinder jackets to the radiator. Each carburetor is a



FIG. 9. LOWER HALF OF CRANK CASE, SHOWING AIR OVERFLOW FOR CARBURATOR

two-jet type, the fuel for the pilot jet passing through a choke tube, somewhat as in the Zenith. The mixture is very accurate, as the air conduct passages through the crank-case from right to left, following the bottom surface of the casing. This warms the air very effectively, but there may be some danger of crank-case fire in case of a blow back into the crankcase. At the same time that the air is warmed in this way, it is circulating to the cooling of the oil in the crank-case.



FIG. 10. UPPER HALF OF CRANK CASE, SHOWING CRANK SHAFT DRIVE FOR MAGNETO

Lubrication System

The oil pump, driven by the lower vertical pump as shown in Fig. 3, forces oil under pressure to all parts of the engine, including the main shaft, as usually demonstrated, a 1/4



FIG. 11. LOWER HALF OF CRANK CASE, SHOWING AIR OVERFLOW FOR CARBURATOR

1/4 inch of oil is required for the main shaft and the connecting rod system.

1/4 inch of oil is required for the main shaft and the connecting rod system.

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1/4 inch of oil is required for the main shaft and the connecting rod system.

is negligibly small. Theoretical experiments by Dr. Prandtl bear out this theory, and demonstrate that the viscous drag varies as $1/V^2$.

Dr. Zahm's experiments on skin friction on the other hand bear down that for even surfaces, bodies covered with weak highly viscous substances as dry varnish, wet varnish, water, oil, etc., all experience the same frictional resistance. It seems therefore reasonably safe to assume that viscous drag is due to internal fluid friction and not to the sliding of the fluid along the surface of the solid.

Density Resistance to a Plane Moving Edgewise

For successfully making vehicles, it has been found that resistance varies as V^2 indicating purely a drag due to shear (Stokes). For small velocity experiments by Allen have shown a resistance varying as $V^{1.5}$ indicating the resistance of viscous drag which we have developed in the preceding paragraphs. But for the vehicles with which we are concerned the resistance of a thin plane surface moving edgewise increases as some higher power of V . This is probably—although it is impossible to state the exact cause—due to the fact that the viscous drag not only exerts translational velocity to the particles which adhere to it in the boundary layer but the boundary layer acts as a source of geysering also by some eddying or rolling velocity to particles adjacent to this boundary layer. It is a commencement of turbulent,

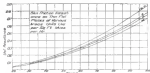


FIG. 13. SKIN FRICTION CHART

eddying motion. As such the extra resistance is proportional to some area A of the body, and to the velocity V , squared, and is termed density resistance and

$$R_d = K A V^2$$

where K is a constant for the fluid

Total Skin Friction. Dr. Zahm's Experiments

$$\text{Total skin friction} = R_v = R_s + R_d$$

R_v = viscous resistance \propto density resistance

Strictly speaking if $R_v \sim V^2$ and $R_d \sim V^2$ as an expression with V raised to a power is not strictly the expression. But for practical purposes the results of Dr. Zahm's valuable experiments have been accepted, his formula being—

$$R = 0.00000125 L^2 V^2$$

where R = resistance for one side of board

L = length in direction of wind in feet

V = velocity in feet per second

In the French Technical Report of the Advisory Committee for Aeronautics, 1913-1915, p. 24, as otherwise a form of equation has been submitted, so as to make the equation consistent with the given value of density resistance:

$$R = 0.0000000125 L^2 V^2$$

where L = area of one side of the board in square feet

except after adjustment. Dr. Zahm's formula may be used to compute the resistance of flat bodies, cylinders, oval bodies when moved to the wind. In Fig. 14, curves of the resistance of plates of various area at varying speed have been plotted, to facilitate such computations.



FIG. 14. ARRANGEMENT OF TEST PLATE IN WIND TUNNEL, DR. ZAHM'S EXPERIMENTS.

Dr. Zahm's skin friction experiments are described in Bulletin, Vol. xiv, pages 287-295, of the Philosophical Society of Washington, June, 1904. The plate was suspended in the wind tunnel as shown in sketch in Fig. 14, with wet shims at either end so as to give purely tangential flow. As the wind friction moved the plate, whenever the displacement was determined by the motion of a sharp pointer attached to one suspension wire and moving over a fine scale lying on the top of the tunnel, and hence the forces were determined. A variety of shapes and surfaces were tested.

Curves for Comparisons with Dr. Zahm's Formula

In Fig. 11, the skin friction resistance in lb. per square foot is plotted against the speed in miles per hour. The resistance increases less rapidly than the area, separate curves have been drawn for several different areas, and the line per unit area on any other surface can be found by interpolation. The curves were plotted by modifying Zahm's formula

$$R_v = K A V^2$$

where K is in feet per second. To show this ratio scale law made it was necessary to multiply by $(\frac{5}{22.3})^2$, or 0.04, giving

$$R_v = 8 \text{ lb. per ft.}^2 V^2$$

In Tables 3 and 5 similar data has been given for speeds in feet per second and miles per hour.

TABLE 3.—SKIN FRICTION RESISTANCE

Speed (feet per second)	1	2	3	4	5	6	7	8	9	10
10	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000
20	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000
30	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000
40	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000
50	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000
60	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000
70	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000
80	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000
90	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000
100	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000

TABLE 4.—SKIN FRICTION FORMULA

$R = 0.0000000125 L^2 V^2$ L = area in square feet

Speed (feet per second)	1	2	3	4	5	6	7	8	9	10
10	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000
20	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000
30	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000
40	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000
50	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000
60	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000
70	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000
80	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000
90	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000
100	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000	0.0000000000

Turbulent Flow, Eddy or Density Resistance

We have already seen in the case of a flat plate normal to the wind that the resistance was due to a region of turbulent, eddying, low pressure behind the plate. This resistance varies as V^2

where A = area in square feet

R = density

V = velocity

It will be assumed for the time being that whenever there is a region of turbulent, eddying flow, there will be a density resistance

$$R \sim V^2$$

A fairly complete demonstration of this has been given by a French author, and will shortly be reproduced in the pages of this paper.

Comparison of Forces Acting Upon Smaller Bodies. The Importance of Kinematic Viscosity and the Reynolds's Number

For the comparison of forces acting on smaller bodies, a knowledge of the kinematic properties and of the wind resistance is essential. The density of the fluid, the viscosity and hence the coefficient of kinematic viscosity, and the compressibility of the fluid all enter into the complete comparison. Consequently, we have seen, may conveniently be applied to all aerodynamic work.

For bodies as small as the resistance is purely of a density or eddy nature—as in the case of a flat plate normal to the wind, and as we shall see subsequently in the case of a wing section at large angles—viscosity does not enter into consideration, or is of so small importance that it may be neglected. In such cases

$$R \sim V^2$$

where A is the area of one face of the plate, comparison between two bodies such as a flat wing and its model become extremely simple.

But for stream line bodies such as struts, cables, wires, and spheres the resistance is compounded of density resistance and viscosity resistance in varying proportions.

Viscosity resistance depends, as we have seen, on velocity, hence dimensions and the coefficient of kinematic viscosity. For such bodies therefore the resistance must be expressed in a form involving these variables, and by the principle of dynamic similarity it can be demonstrated that

$$R = \rho V^2 L^2 f\left(\frac{V}{\nu}\right)$$

where f is some unknown function and V is of the same dimensions as ν . The $V^2 L^2$ leaves out the density resistance,

$f\left(\frac{V}{\nu}\right)$ the viscosity resistance, where $\frac{V}{\nu}$ is the Reynolds's number, ν the kinematic viscosity.

$$R = \rho V^2 L^2 f\left(\frac{V}{\nu}\right)$$

The reader will now appreciate the importance of the Reynolds's number in comparing the resistance forces on the above mentioned bodies.

It is quite incorrect to compare such bodies, making allowance for variation in V and L only, unless the Reynolds's number is the same for the two bodies under comparison. In practice it is very rare that comparison of forces are made with reference to two different fluids. We are almost always concerned with bodies in air. The coefficient of viscosity becomes a constant, and instead of considering the Reynolds's number, we can drop the ν and compare bodies having the same product $V L$.

Stream Line Bodies

A stream line body may be defined as any which has a gradual change of curvature along any section, and which



FIG. 15. STREAM LINE BODY (Stream Line Body)

when moved through air or water at ordinary speeds makes little disturbance or turbulent wake. Such a body may sit in a stream without appreciable aerodynamic resistance.

Energy Considerations for a Perfect Fluid Flowing Past a Stream Line Body

It is most useful to have a definite idea of the exchange of energy which occurs in such a case. The first treatment appears to have been given by W. Strouhal. Consider the fluid in the vicinity of the body to be divided up into a large number of imaginary tubes of flow. Well ahead of the body where the stream is as yet undisturbed the energy of the fluid will be that due to the static pressure p_s of the stream and the kinetic energy head of V , the undisturbed velocity. In a perfect fluid this will remain a constant along any tube of flow in Darcy's theorem, and is equal to

$$\frac{p_s}{\rho} + \frac{V^2}{2} = \text{constant}$$

For the position A as shown in Fig. 15A of the body, the total



FIG. 16. STREAM LINE BODY (Stream Line Body)

of flow wakes out, the velocity and the kinetic energy level decrease and the pressure on the body becomes greater than the static pressure p_0 . The most of the body therefore does work upon the fluid in contact with it. This is also evident by considering the effect of curvature and the centrifugal force resulting from it. For the portion M the forces round together, the velocity increases and the body is under the action of a pressure less than p_0 —it reacts under suction and the fluid does work on the body. By similar reasoning it can be shown that the portion N the body works upon the fluid, and for the portion P , the fluid works upon the body. The balance of work done as the body is thus found to be zero.

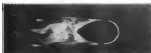


FIG. 17. FLOW OVER A SMOOTH BODY.

Stream Line Bodies in a Viscous Fluid

At slow speeds in water almost perfect stream line motion has been observed and recorded by Dr. Adkins (see Fig. 18). But at ordinary speeds—such as those for boats—there is



FIG. 18. FLOW OVER FISH BODY.

always a region of turbulence and eddy motion such as we have already observed in the case of the flat plate, motion caused by a surface of discontinuity between the main stream and the turbulent region. The eddy motion arises in part due to pressure differences in the undisturbed stream and the region behind the body, in part due to viscosity. The exact theoretical or mathematical of the nature of flow are unimportant

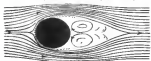


FIG. 19. FLOW AROUND A CYLINDER.

from the designer's point of view. It is more important to notice that such as in the case of the flat plate, this turbulent region will be a region of low pressure and will introduce a drag or eddy resistance.

The density resistance for a stream line body may be said to increase with the extent of the turbulent region. Thus in

Figs. 17 and 18 depicting two standard forms, the first also has a smaller turbulent region and consequently less resistance. On the other hand, as the thickness ratio, or the ratio of length to maximum thickness, of a stream line body increases, it is seen in shear and the velocity drag increases also, the same ratio must be kept a given reasonable limits even from a purely aerodynamic point of view.

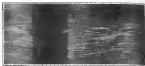


FIG. 20. FLOW OVER A CANNON WING AT 2°.

Resistance of Wires, Cables and Cylinders
Fig. 19 represents (approximately) the third standard form a cylindrical body, such as a wire or cable, of most aerodynamic shape.

It is also seen that the resistance will be partly due to viscosity in the front part of the cylinder and partly due to eddy or density resistance. The forces in action will therefore be represented as part only stated by an expression of the form

$$C_d \rho V^2 A$$

And two wires or cables will each be comparable when the same for both or simply when the product CA is the same.

Fluid Motion Around Wing Surfaces

It is to Langley, where all other wings that we see an approximation of the value of rounded surfaces. A good wing section may give a lift/draft ratio of 18 as compared to 10 or 7 of a flat plate, and it is the remarkable efficiency of a wing surface which has largely revolutionized aviation.

In wing surfaces, in comparison to standard types of flat. For the small angles up to 3° or thereabouts a steady flow is shown in Fig. 20 for a typical airplane wing. At this angle—often termed the limit of laminar flow—turbulence begins. At 16°, as shown in Fig. 21, the turbulence is again considerable. Finally a second critical or "stall point" is reached at 18° for the same wing. Here an extremely turbulent type of



FIG. 21. FLOW OVER A CANNON WING AT 16°.

motion, as shown in Fig. 22 is found, and the lift of a wing section is reduced. Beyond this "stall point" the motion becomes extremely unsteady and the lift decreases.

The lift of a wing, as expressed above, varies directly as C_d with a different coefficient for every angle of incidence. Where turbulent flow is present this is readily explainable.

as the case of the flat plate, on the hypothesis of low pressure region at the back of the wing.

It is the lifting power at small angles and is a condition of steady flow that offers theoretical difficulties. The most fully explanation is offered by Kutta's theory or the vortex theory of circulation. We shall reserve the full treatment of this theory also to a special article, restricting ourselves to the latest outlines.



FIG. 22. FLOW OVER A CANNON WING AT 18°.

As a surface wing in steady flow gives of a series of trailing vortices as depicted diagrammatically in Fig. 23, these vortices are constantly destroyed and renewed. The vortices arise in these vortices and their interaction is such that—the hydrodynamic theory demonstrates—they have a downward momentum, and active and motion being equal, an upward wing motion an upward momentum.

The lift or drag of a wing is for all practical purposes also as varying directly with C_d with a different coefficient for every angle of incidence.

At high angles of incidence, the lift is almost entirely a component of the density resistance, and we see that what is due to the case in practice, is also theoretically correct. But at small angles and steady flow the resistance is more of a viscous nature, more akin to skin friction. And this

FIG. 23. AIRPLANE WING WITH TRAILING VORTICES.

factor, as we have already seen, varies as V^2 and depends also on the dimensions of the body. This introduces considerable difficulties, as we shall see later, in comparing resistance in actual flight from small model experiments at low speeds.

As to the form of wing giving the best results, no general law is yet available, and each type of wing must be considered separately.

This section considers but a brief introduction to aerodynamic theory, but will perhaps assist the reader in the appreciation of the extensive aerodynamic data which we shall present later.

References to Authorities for Sections 2 and 3

A Review of Hydrodynamic Theory as Applied to Engineering Aerodynamics, J. C. DUMAS, International Engineering Congress, San Francisco, Sept., 1935. An authori-

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Photographic Investigation of the Flow Round a Model Aeroball, KOLLE. Dutch report, 1931-1932, No. 38.

These papers in the British report contain some beautiful and instructive photographs.

Physics Notes

By W. H. BROS. PH.D.

DISCOVERIES OF PAUL HERMAN PAIR A NEW PLANE WITH SPECIAL REFERENCE TO AERODYNAMICS. G. R. BROS. and Robert Jones (Proc. Roy. Soc., 1935, 100-100). The investigation is based on the Kirchhoff-Helmholtz theory of discontinuous stream line motion of a fluid moving through an orifice or past a barrier. Integrals are obtained giving the lengths of the and the shape of the plane. It is shown how these integrals may be evaluated easily, but the reduced expressions are so large that they cannot well be applied to special cases. By means of approximate methods shorter reduced values are obtained. These results are used to find the lift and drag on the surface of planes. The relative advantages of planes having different shapes and along different lines are also considered. The numerical results are set strictly quantitatively correct, since the fluid is assumed inviscid and incompressible and the motion is supposed to be two dimensional, which assumptions are not true for actual bodies. Additional points in planes of more than one level are also considered. The following conclusions are drawn from the quantitative results: Both lift and drag are large when the angle between the plane to largest. This condition, points to the fact that the ratio of lift to the area of the plane increases with the angle of bend. The ratio of lift to drag is greatest when the angle between the plane is large. It is also seen that as the length of the front plate increases the total lift decreases, whereas the ratio of lift to drag increases. Combining the same conclusions, a plane with a large bend close to the front edge gives the best ratio of lift to drag, while a fairly flat plane bent near the rear edge gives the best ratio of lift to drag. All these conclusions show that a distinct advantage is gained by using a curved plane.

THE USE OF COILS OF CABLES IN A STREAM OR AIR, J. A. HUGHES. (Proc. Roy. Soc., 1935, 100-100). Cylindrical tubes of different diameters were placed in a stream of air (produced in a fan, the objects of the wind tunnel) and were measured by experiment with a pressure gauge. Stream was passed through the tubes and the lift was measured in the weight of condensed steam. Conclusions show that at all velocities the lift of a coil of cable is greater than that of a straight cable of the same length and diameter. The lift is also greater than that of a coil of cable of the same length and diameter. The lift is also greater when the bend and fixed the wind.

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